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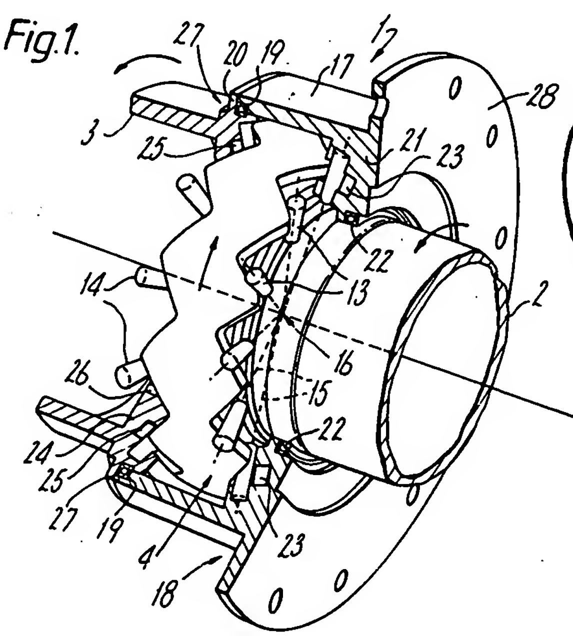
## (54) Improvements in or relating to gear


(57) An annular reduction gearbox suitable for a rotable antenna comprises a double-sided crown wheel cam 4 with a saw-toothed cam profile cut on both sides. The crown wheel cam 4 is mounted on the input drive shaft 2 at a small angle to the axis of the input drive shaft such that it can rotate freely. On each side of the crown wheel cam 4 there is a ring of equispaced rollers 13, 14 whose rotational axis meet at the point of intersection 16 of the central plane of the crown wheel cam 4 and the axis of the gearbox. The input set of rollers 13 is spatially fixed, being located in a gearbox housing member 28, while the second set is located in an end plate 24 provided on the output shaft 3 which is coaxial with the input shaft. As the input drive shaft rotates the crown wheel cam performs a swash-plate motion and the action of the input rollers 13 on the complementary cam profile causes a rotational movement to be superimposed on the swash-plate motion. These crown wheel cam movements interact with the output shaft cam rollers 14 to cause rotation of the output shaft. The overall gearing is selected by choosing the ratios of number of rollers to member of cam profiles on the two sides of the crown wheel cam.

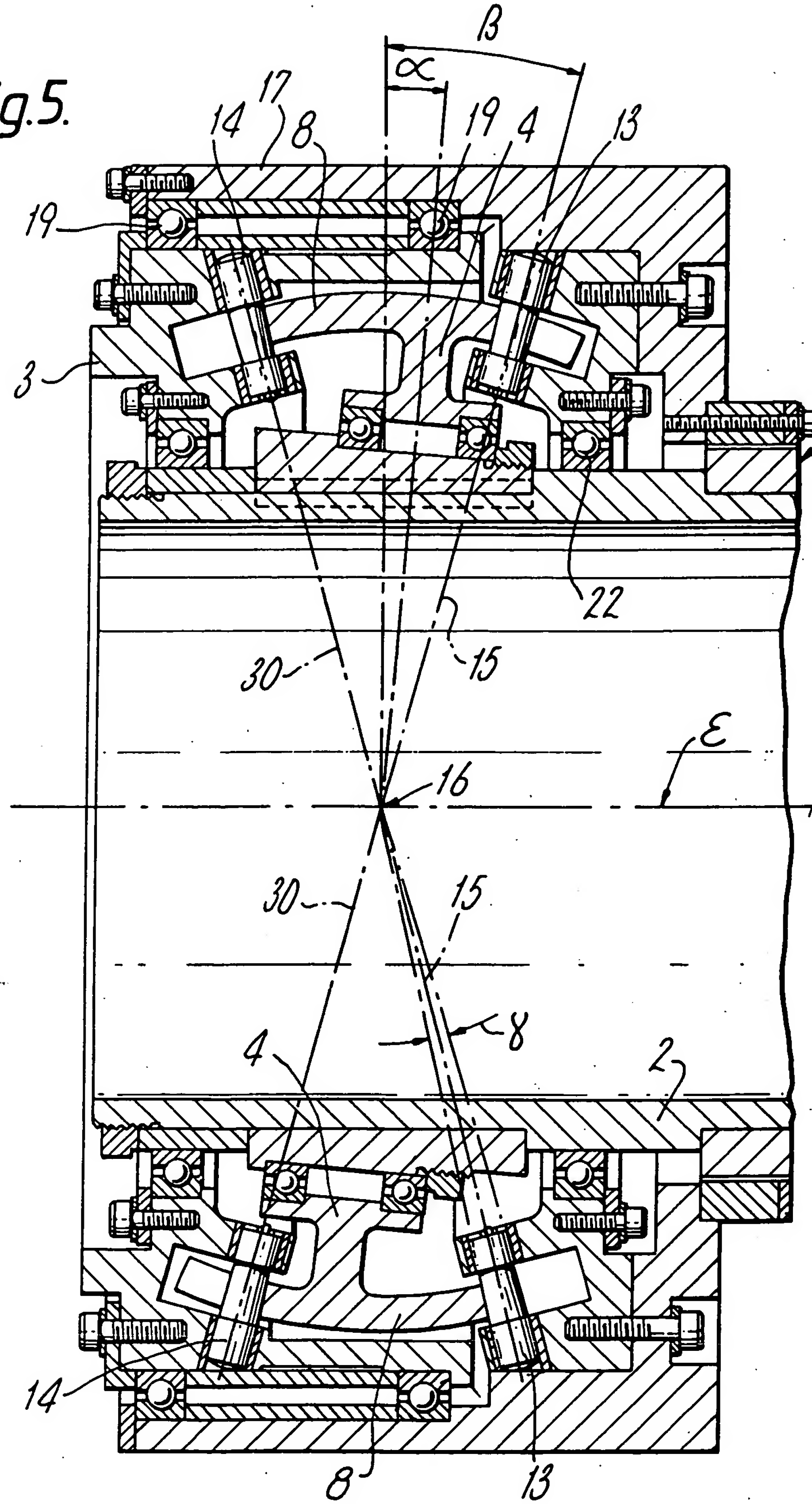

Fig.1.




475/164

GB 2 117 474 A

Fig.5.



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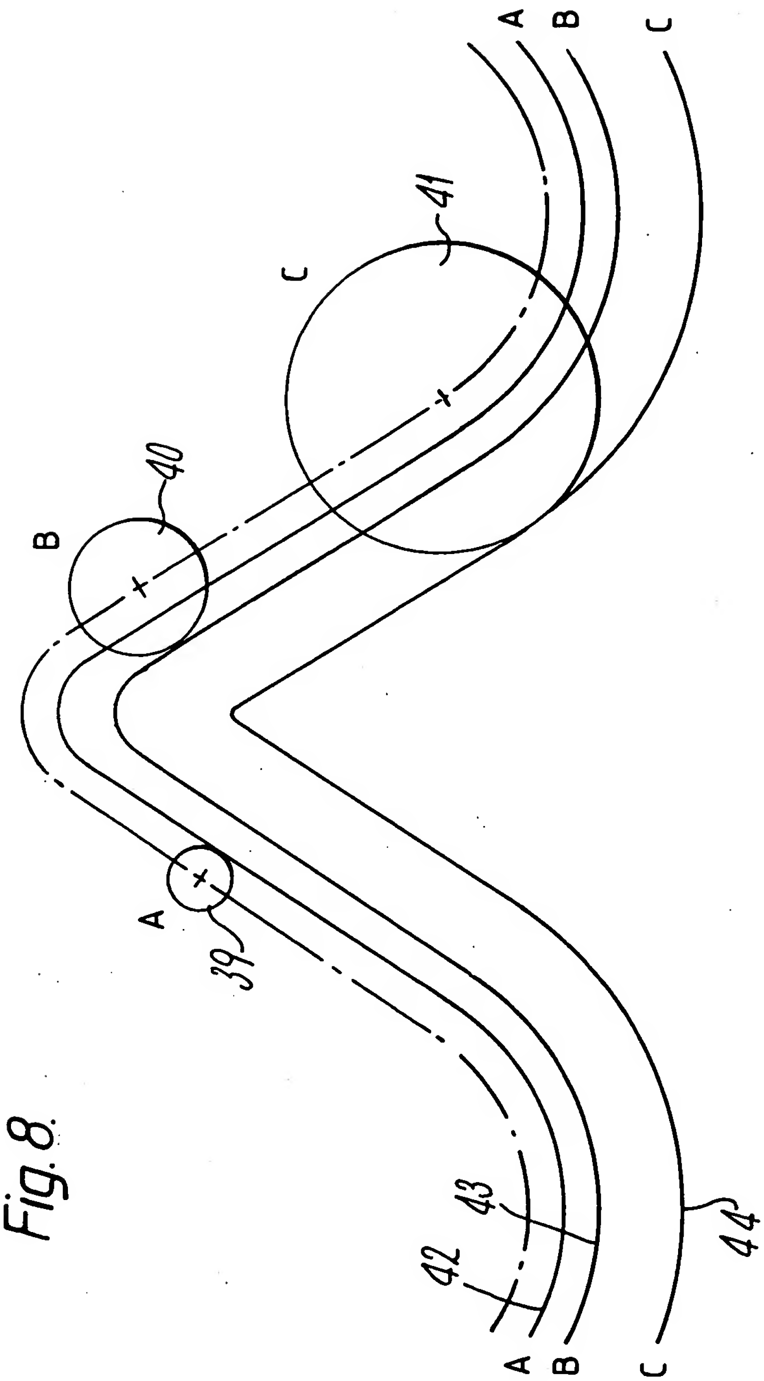


Fig. 8

The invention will now be described by way of example only with reference to the accompanying drawings of which:

*Figure 1* is a perspective view, partly cut away, of a crown wheel cam gear box according to the invention;

*Figure 2* is a schematic drawing showing the double-sided crown cam of the gear box;

*Figure 3* graphically illustrates the disposition of the axis of rotation of the crown wheel cam; and

*Figure 4* is an exploded view of the crown wheel cam mounting in the gear box.

*Figure 5* shows a cross-section through the gear box;

*Figures 6A-6F* show the effect on the cam profile of changing the throw angle of the crown wheel;

*Figure 7* shows the effect of changing the cone angle of the rollers; and

*Figure 8* illustrates the effect of changing the angle of taper of the rollers.

The crown wheel cam gear box 1 shown in Figures 1 to 4 is used to connect a cylindrical input drive shaft 2 to a cylindrical output shaft 3. Rotation of the input drive shaft 2 is translated to the output shaft 3 via a double-sided crown wheel cam 4. A saw-tooth cam profile 5, 6 is cut on both sides of the wheel 4. The crown wheel 4 is mounted such that its axis 7 is set at an angle of 5° to the main centre axis $\varepsilon$ of th gear box assembly.

The mounting of the crown wheel 4 is shown in Figure 4. The crown wheel 4 comprises a cylindrical portion 8 into the axial ends of which the cam profiles 5 and 6 are cut and an inward radially extending web or disc portion 9. At the inward extremity of the web 9 a double ball race bearing 10 cooperates with complementary grooved annular bearing members 11 and 12 fixed to the input shaft 2 to allow the crown wheel to rotate freely in its inclined plane.

On each side of the crown wheel cam 4 there is a ring of equi-spaced tapered rollers 13 and 14 many of which are in contact with the cam surfaces 5 and 6. The axes 15 of the input rollers 13 and the axes of the output rollers 14 (not shown for clarity) lie on the surfaces of very shallow cones whose apices meet at the point of intersection 16 of the plane of the crown wheel cam 4 and the main centre axes $\varepsilon$ of the assembly. Surrounding the crown wheel cam 4 is a cylindrical portion 17 of a gear box casement member 18 which is provided at one end with a bearing 19 for cooperation with a complementary recess 20 provided in the end of the output shaft 3 and at the other end with an inwardly extending radial wall 21 with a bearing 22 within which the input shaft 2 rotates. The output side of the wall 21 is provided with bearing seats for location of the input rollers 13 and has an annular recess 23 into which the cam profile 5 can pass allowing the input rollers 13 to follow the cam profile 5. In similar manner the output shaft 3 also has a radial wall portion 24 provided with an annular groove 25 for access by the cam profile 6 and bearing seats for location of the output rollers 14. A circular bearing 26 is provided on the inner edge of the output shaft wall portion 24 for rotatable mounting on the input shaft 2. Extending radially outwards from the output shaft 3 is a wall portion 27 to provide the recess 20 for engagement with the bearing 19 in the casement member 18. The casement member 18 is provided with an outward flange 28 for attachment to the remainder of the gear box housing (not shown).

The gear box operation will now be explained by considering the effect of a rotation of the input shaft 2, as shown in Figure 1, which rotates freely in the bearings 22 and 26. Looking at the right hand side of the Figure, as the input shaft rotates about the main centre axes $\varepsilon$ it carries with it the inner members 11 and 12 of the crown wheel cam bearings 10. This causes the plane of the crown wheel cam 4 to perform a swash-plate type movement. If the number of cycles on the cam profile 5 is one less than the number of input rollers 13 and these are spatially fixed to the gear box housing, then as the cam profile 5 rides over each roller 13 in turn, the crown wheel cam 4 rotates slowly backwards on its own inclined bearings as indicated in Figure 1 by an amount equal to one cam cycle for each 360° rotation of the input shaft 2. A gear ratio is thus obtained between the input shaft 2 and the crown wheel cam 4. The crown wheel cam rotation however, is about the inclined axis 7.

Looking now at the left hand or output side of the gear box where a similar arrangement is used except that the output rollers 14 are now attached to the ouput shaft 3. Again the inclined crown wheel cam 4 provides the same swash-plate motion but there is also a slow rotation superimposed. By again adopting a difference between the number of cam cycles in the profile 6 and the number of output rollers 14, a second gear ratio can be obtained between the crown wheel cam and the output shaft and in addition the axis of the output rotation is now realigned with the axis of the input shaft, ie the main centre axis $\varepsilon$. By choosing suitable members of rollers and cam cycles for the two sides of the gear box a wide range of different gear ratios can be obtained, varying from 4:1 to 400:1 in the same basic size and shape of box.

Figure 5 shows an arrangement of the crown wheel cam gearbox to illustrate the practical considerations involved in implementing the invention. An annular torque motor 29 is mounted on the input shaft 2. The axis of rotation of the crown wheel 4 is set at a throw angle of 5°. The axes 15 of rotation of the spatially fixed input rollers 13 are inclined at a cone angle $\beta$ of 15° so as to intercept the axes 30 of rotation of the output rollers 14 at the point 16 on the central axis $\varepsilon$ of the gear box. The cylindrical portion 8 of the crown wheel 4 is made part-spherical such that the cam profiles 5 and 6 engaging the rollers 13 and 14 are cut normal to the cam portion 8. The angle of taper of the cam rollers is shown by the angle $\gamma$. In this arrangement two axially separated bearings 19 are provided between the output shaft 3 and the casement wall 17 of the gearbox housing. Other parts common to the arrangement shown in Figure 1 are labelled with like reference numerals.

The gearbox operates by the transfer of power between rollers and cam profiles. Because of this the

designs of the rollers and the cam profiles are very important, and the influence of various gearbox parameters must be considered. The important parameters are as follows:

a) the number of cam profiles and rollers within the gearbox;

b) the throw angle $\alpha$ ie the angle between the axis of rotation of the crown wheel and the axis of the input shaft;

c) the cone angle $\beta$ ie the angle between the axis of rotation of each roller and the plane perpendicular to the input shaft axis;

d) the roller angle $\gamma$ ie the angle at which the surfaces of the roller taper; and

e) the radius of the crown wheel and roller assemblies.

a) *The number of cams and rollers*

The gearbox ratio is governed by the numbers of cams and rollers and can be shown to be represented by the formula:

$$\text{Gear Ratio} = \frac{1}{\left(1-\dfrac{N_1N_3}{N2N4}\right)} : 1$$

where $N_1$ and $N_2$ are the respective number of rollers and the number of cam profiles on the input side and $N_4$ and $N_3$ are the respective number of rollers and cam profiles on the output side.

In general the greater the N values, the higher is the gear ratio. By having the number of cam profiles equal to one less than the number of complementary rollers, several rollers will be in contact with their respective cams at any one time. This has the advantage that the load, due to the transmitted power is shared between a number of rollers, making the load on an individual roller low.

b) *The throw angle*

The throw angle $\alpha$ is the dominant influence on the amplitude or height of the cam profile: the greater the throw angle the greater the height.

Once the number of cam profiles, and hence the cam's base length, has been established, the throw angle must be suitably chosen.

Figures 6A-6F show the effect of increasing the throw angle on the cam profile, for a cam of fixed base length L. The increase of throw angle is represented by the increase of cam height a. In each case the locus 31 of a roller 32 is shown.

With the smaller throw angles as shown in Figure 6A and 6B the resulting shallow cam flank angle 33 causes the loads on the roller bearing to be large. By increasing the throw angle (ie the height a), the cam flank becomes steeper and, for the same power transmitted through the gearbox, the roller bearing loads are reduced. With further increase of the throw angle, the upper crest 34 of the cam profile becomes pointed and the roller begins to loose contact with the cam profile in the region of the upper crest 34 as shown by the portion 35 of the loans 31 of the roller in Figures 6E and 6F. The effect of lo sing cam contact is to reduce the power capacity of the gear box. In addition, the more pointed the cam profile is made the structurally weaker it becomes and the maximum loading must therefore be reduced. An optiumum range of throw angle is thought to be as illustrated in Figures 6D and 6E. This gives a cam flank angle 33 of about 45°.

It is advantageous to have as small a throw angle as the above considerations permit. This is so that the out-of-balance forces, due to the inclination of the crown wheel, are minimised. These forces can be exactly balanced by counterweight masses attached to the input shaft, however, with small throw angles only small masses are required and hence the overall weight of the gearbox is kept low.

c) *The cone angle*

The cone angle has an effect on the cam profile which can best be seen by considering the spherical geometry of the crown wheel.

Figure 7 shows cam profiles 36-38 for cone angles of 15°, 45° and 70°. In each case the number of cam profiles and the throw angle are constant. As can be seen the effect of increasing the cone angle is to sharpen the peaks of the cam profiles. Thus to avoid a reduction in power capacity due to the strength limitation of a sharp cam it is preferable to select as low a cone angle as possible.

Increasing the cone angle also reduces the clear space through the gearbox as well as increasing the mass of the crown wheel. These factors again favour a low cone angle. A lower limit exists for the cone angle since the troughs of the cam profile must be clear of the crown wheel boss. A cone angle of about 15° has been found preferable.

d) *Roller angle*

The roller angle has a large influence on the form of the cam profile as shown in Figure 8. Three rollers 39,

40 and 41 of increasing roller angle are drawn with their associated cam profiles 42-44. By increasing the roller angle a narrower cam profile results which could reduce power capacity due to low cam strength. In the extreme case of a very large roller angle, the additional problem of the roller losing contact with the cam profile comes into play. The consequence of this, as before, would be a further reduction in power capacity. In general, therefore, too large a roller angle leads to problems with the cam. Using the roller 39 with a small angle leads to problems due to the roller itself. The roller is subject to high Hertzian stresses which could lead to fatigue problems. The shear strength of the roller is reduced when the roller angle is smaller thereby limiting power capacity. In addition the rotational speed of the roller increases with smaller roller angles and this increases fatigue effects and may also increase the friction associated with the support bearings for the roller. A compromise roller angle as shown for the roller 40 is considered to be optimal.

e) *Radius of the crown wheel and roller assemblies*

To minimise the loading between the rollers and the cams it is desirable to have the radius of the crown wheel as large as possible within any existing limitations of space available for the gearbox. With a large crown wheel full advantage can be taken of a clear space provided through an annular form of the gearbox. In addition the drive stiffness will be increased relative to that of a smaller crown wheel and backlash reduced.

On the other hand the bearings and shaft seals for a large radius crown wheel and roller assembly will cause more frictional losses than for one of smaller radius. These losses, however, should not be very great.

The invention can also be applied to a single-sided crown wheel cam with the output being taken directly from the crown wheel cam. This would not, however, produce an output axis of rotation in alignement with the input rotational axis and is therefore not desirable. To overcome this misalignment, rollers may be provided in an end plate of an axially aligned output shaft, the rollers engaging oval slots provided in the crown wheel. Thus the off-axis rotation of the crown wheel is used to drive the aligned output shaft. In a double-sided crown wheel cam the gearing of the two sides of the crown wheel cam need not be in the same sense, since by varying the ratios of cam cycles to rollers the second gear ratio can be made to add or subtract from the first ratio. In all cases, however, the invention involves swash-plate movements and when the components are made in annular form the diameter of the passage available for cables, waveguides etc is equal to the internal diameter of the input drive shaft 2.

The advantages of the present invention are as follows:

a) the annular form of construction allows space through the centre for cables and/or waveguides;

b) the gear box has a high load and shock carrying capacity since a number of rollers are in contact with the crown wheel cam and are driving all the time, and the cam shapes have an inherently high strength;

c) the gear box is very efficient with all parts being in rolling contact only and the mating surfaces all being projected from a single point on the axis of the gear box;

d) the gear box is small, compact and light weight for a given ratio and torque capacity;

e) the gear box is very reliable with a long life since wear is minimised as there are no sliding parts;

f) a wide range of gear ratios is available for the same basic size gear box;

g) there is low referred inertia since the input shaft is the only part rotating at the input speed - an important consideration in many servo-controlled systems; and

h) there is a very high inherent "stiffness" as would be useful for servo applications, since there are not a lot of resilient gear wheels and shafts as in conventional gear boxes.

It will be apparent to those skilled in the art that many modifications are possible to the described embodiment, all falling within the scope of the invention described.

## CLAIMS

1. A gear box having:

a) an input shaft;

b) an output shaft; and

c) a crown wheel cam operatively connecting the input shaft to the output shaft via at least one set of cam followers,

wherein the axes of the input shaft and the output shaft are substantially in alignment and perpendicular to the plane of the crown wheel and the axis of rotation of the crown wheel is set at a small angle to the axis of rotation of the input shaft.

2. A gear box as claimed in claim 1 wherein the crown wheel cam is rotatably mounted on the input shaft.

3. A gear box as claimed in claim 1 or 2 wherein one side of the crown wheel cam is provided with a first cam surace such that rotational movement of the input shaft is conveyed to the crown wheel via a first cam action.

4. A gear box as claimed in claim 3 wherein the crown wheel cam comprises a circular disc having around its circumference a substantially cylindrical portion formed with the first cam surface on an axial end thereof.

5. A gear box as claimed in any one of claims 2 to 4 wherein the first cam surface is engaged by a circular array of spaced input pins or rollers each at a fixed radial distance in a plane normal to the axis of the gear box.

6. A gear box as claimed in claim 5 wherein the input pins or rollers are located in a gear box housing member.

7. A gear box as claimed in any one preceding claim wherein there are provided slots in the output side of the crown wheel cam for engagement by rollers fixed in an end plate provided on the output shaft.

8. A gear box as claimed in any one of claims 3 to 6 wherein the other side of the crown wheel cam is provided with a second cam surface whereby rotational movement is conveyed to the output shaft via a second cam action.

9. A gear box as claimed in any one of claims 3 to 8 wherein the or each cam surface comprises a plurality of identical cam profiles so as to form substantially a saw-tooth arrangement.

10. A gear box as claimed in claim 8 or 9 wherein the second cam surface is provided on the other axial end of the cylindrical portion of the crown wheel cam.

11. A gear box as claimed in claim 9 or 10 wherein the angle of offset of the axis of the crown wheel cam to the axis of the input shaft is arranged such that the cam flank angle is approximately 45°, where the cam flank angle is defined as the angle which the flank of the or each cam profile makes with the plane of location of the input pins or rollers.

12. A gear box as claimed in any one of claims 8 to 11 wherein the output shaft is provided with an end plate, there being located in the end plate a second circular array of spaced output pins or rollers for engagement with the second cam surface.

13. A gear box as claimed in any one of claims 5 to 12 wherein the axes of rotation of the pins and rollers fall on one of the surfaces of two shallow cones whose apexes meet at the axial centre of the cam wheel.

14. A gear box as claimed in claim 13 wherein the surfaces are preferably inclined at an angle of about 15° from the plane which is perpendicular to the axis of the gear box and passes through the point of intersection of the apices.

15. A gear box as claimed in any one of claims 9 to 14 wherein the number of cam profiles is equal to one less than the number of complementary pins or rollers.

16. A gear box suitable for use with a rotatable antenna as claimed in any one preceding claim wherein the input shaft, the output shaft and the crown wheel are annularly formed whereby cables and/or waveguides can pass through the axial passage formed thereby.

17. A gear box as claimed in claim 16 wherein the input shaft is driven by an annular torque motor.

18. A gear box substantially as described with reference to Figures 1 to 8 of the accompanying Drawings.

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